MESHING HELICAL ROTORS

This application claims the benefit of the filing date of provisional application 60/433,720, having a filing date of December 16, 2003.

Background of the Invention

Screw compressors and expanders are composed of meshing screw or helical rotors. As in the case of gears, screw rotors have pitch circles which represent locations of equal tangential velocity for conjugate pairs of rotors. The spiral grooves in the rotors are the locations of the volumes of gas which are trapped and in the case of compressors, compressed due to the coaction of a conjugate pair of rotors and an enclosing casing. Accordingly, the volumes of the spiral grooves are a major design consideration, their width, depth, length and number being important design variables. The shape of the cross section of the spiral grooves includes the variables of width and depth as well as the shape requirements for the driving/driven coaction between the conjugate pair of rotors. Additionally, the conjugate pair must meet the sealing requirements as the line contact advances along the rotor profile in the driving/driven coaction and as the rotor tips and end faces coact with the enclosing casing. This line contact follows the perimeters of the rotor profiles and is therefore at a varying tangential speed and has significant radial components. Additionally, the shape and cross section of the spiral grooves must meet requirements for ease of manufacture and cutting tool life. One problem associated with conventional screw rotor designs is that rotor profiles have generally been designed using a point generated and or circular profiles. These types of profiles are generally more difficult to machine as well as exposing the rotors to more significant impact with respect to seal line length, drive band contact stress, service life, and sensitivity to temperature fluctuations.

There exists a need therefore for a screw rotor profile for reducing seal line length, reducing contact stress, increasing service life, and exhibiting more flexibility to temperature fluctuation.

Summary of the Invention

It is an object of this invention to increase the efficiency and longevity of a screw machine.

It is another object of this invention to provide a screw rotor profiles having reduced blow-hole area for improved efficiency.

It is yet another object of this invention to provide improved rotor tip curves which are less sensitive to tip clearance modification and which can be used for tapered rotors.

It is a further object of this invention to achieve the disclosed performance based objects while improving the manufacturability of the screw rotor profiles.

Another object is to reduce the contact stress between the male and the female rotors of a screw machine.

These objects, and others as will become apparent hereinafter, are accomplished by the present invention, a conjugate pair of intermeshing rotors having helical lobes having helical crests and intervening grooves and adapted for rotation about parallel axes within a working space of a screw rotor machine. Each rotor has a tip circle, a pitch circle, and a root circle, one rotor of each pair being a female rotor such that the major portion of each lobe of said female rotor is located inside the pitch circle of the female rotor. The other rotor is a male rotor formed such that the major portion of each lobe of said male rotor is located outside said pitch circle of the male rotor. The lobes of one rotor follow the grooves of the other rotor to form a continuous sealing line between said pair of rotors, each of the lobes having a primary and secondary flank portion, wherein the primary flank portion of said lobes of the female rotor have a profile formed from at least one ellipse and the primary flank

portion of the lobes of the male rotor have a profile formed from at least one ellipses.

Brief Description of the Drawings

For a fuller understanding of the present invention, reference should now be made to the following detailed description thereof taken in conjunction with the accompanying drawing wherein:

FIG. 1 is a simplified transverse section through the rotors of a screw machine employing the present invention.

FIG. 2 is a simplified view of the blow hole of the present invention as compared to the prior art.

Description of the Preferred Embodiments

In the Figure 1, the numeral 10 generally indicates a screw machine such as a screw compressor or an expander. Screw machine 10 has a casing 12 with overlapping bores 12-1 and 12-2 located therein. Female rotor 14 has a pitch circle, P_F, and is located in bore 12-1. Male rotor 16 has a pitch circle, P_M, and is located in bore 12-2. The axes indicated by points A and B are perpendicular to the plane of Figure 1 and are parallel to each other and are separated by a distance equal to the sum of the radius, R_F, of the pitch circle, P_F, of female rotor 14 and the radius, R_M, of the pitch circle, P_M, of male rotor 16. The axis indicated by point A is the axis of rotation of female rotor 14 and the center of bore 12-1 whose diameter generally corresponds to the diameter of the tip circle, T_F, of female rotor 14. Similarly, the axis indicated by point B is the axis of rotation of male rotor 16 and the center of bore 12-2 whose diameter generally corresponds to the diameter of the tip circle, T_M, of male rotor 16.

As illustrated, female rotor 14 has six lands, 14-1, separated by six grooves, 14-2, while male rotor 16 has five lands, 16-1, separated by five grooves 16-2.

Accordingly, the rotational speed of rotor 16 will be 6/5 or 120% of that of rotor 14. Either the female rotor 14 or the male rotor 16 may be connected to a prime mover (not illustrated) and serve as the driving rotor. Other combinations of the number of female and male lands and grooves may also be used.

Generally referring to FIG. 1, the major portions of the rotor profile, that is leading flank or secondary flank D-B for both the male and female rotors, trailing or primary flank A-E for both the male and female rotors of the female and male rotors, 14 and 16, of the present invention are different ellipses or generated by different ellipses, with the tip or root portions being circular arcs. The ellipse allows for a continuously changing curved profile, as opposed to a fixed profile with circular curves, yielding a high radius at the drive band for reduced contact stress on the drive band, and a low radius near the rotor tip.

With reference to the FIG. 1, the male rotor tip segment A_M-B_M and the female rotor root segment A_F-B_F are each circular arcs having their centers at pitch point P_M and P_F, respectively. The male rotor tip circle has a tangent contact point with segment A_M-B_M between points A_M and B_M. The female rotor bottom circle with the root diameter of the female rotor has a tangent contact point with segment A_F-B_F between points A_F and B_F. The male rotor tip segment A_M-B_M allow the male tip to have the traditional seal strips or to have the tapered rotors should they are required.

The leading flanks or secondary flanks of the male and female rotors have two segments. Convex segment B_M-C_M is part of an ellipse, with one of its axis overlapped with line B_M-P_M and having a common tangent at B_M with A_M-B_M. Concave or concave-convex segment B_F-C_F is conjugally generated by ellipse B_M-C_M. Segment B_F-C_F has a common tangent at B_F with circular arc A_F-B_F. Points C_M and C_F may be just on or inside or outside the pitch circles of the male and female rotors respectively. Convex segment C_F-D_{1F} is part of an ellipse, with one of its axis overlapped with the radius of D_F-D_{1F} at point D_F, and which has a common tangent at C_F with B_F-C_F and has a common tangent at D_{1F} with the circular arc D_F-D_{1F}. Concave

segment C_M-D_{1M} at the male rotor leading flank is conjugally generated by ellipse C_F-D_{1F}. Segment C_M-D_{1M} has a common tangent at C_M with B_M-C_M, and has a common tangent at D_{1M} with circular arc D_M-D_{1M}.

The tip portion of the female rotor and the root portion of the male rotor have two segments. Segments D_M-D_{1M} and E_M-D_M are the two segments of the root portion of the male rotor, and segments D_F-D_{1F} and E_F-D_F are the two segments of the tip portion of the female rotor. D_M-D_{1M} is a concave circular arc with its center on the pitch circle of the male rotor, and D_F-D_{1F} is a convex circular arc with its center on the pitch circle of the female rotor. E_M-D_M is a convex circular arc with its center at the center A of the male rotor, and E_F-D_F is a convex circular arc with its center at the center B of the female rotor. Segment D_M-D_{1M} has a common tangent at D_M with E_M-D_M, and segment D_F-D_{1F} has a common tangent at D_F with E_F-D_F. The female rotor tip segments allow the female tip to have the traditional seal strips or to have the tapered rotors should they are required. The male root segments allow the male root to have the traditional seal grooves.

The trailing primary flanks of the male and female rotors have two segments. Segments A_M-F_M and F_M-E_M are the two segments of the trailing flank of the male rotor, and segments A_F-F_F and F_F-E_F are the two segments of the trailing flank of the female rotor. Convex segment A_M-F_M is part of an ellipse, with one of its axis overlapped with line A_M-P_M and having a common tangent at A_M with A_M-B_M. Concave segment A_F-F_F is conjugally generated by ellipse A_M-F_M. Segment A_F-F_F has a common tangent at A_F with circular arc A_F-B_F. Point F_F is inside the pitch circle of the female rotor. Convex segment F_F-E_F is part of an ellipse, with one of its axis overlapped with the radius E_F-O_2 at point E_F, and which has a common tangent at F_F with A_F-F_F and has a common tangent at E_F with the circular arc E_F-D_F. Convex-concave segment F_M-E_M at the male rotor leading flank is conjugally generated by ellipse F_F-E_F. Segment F_M-E_M has a common tangent at F_M with A_M-F_M, and has a common tangent at E_M with circular arc E_M-D_M.

As illustrated in FIG. 2, as a consequence of the above described profile, the area of the blow hole formed by the tip and leading flank sections of the meshing female 14 and male 16 rotors, is reduced by its shape being curved and narrower, in comparison to prior art blow holes formed by non-elliptical profiles, without reducing the blow hole height. By avoiding reduction in height, reasonable gas torque is maintained from the male to female rotor. As known in the art, the blow hole is a leakage channel which connects the leading and following cavities, and it reduces the total efficiency of helical screw compressor. This design, as described and as shown in FIG. 2, has the advantage of increasing performance of the compressor.

As a further consequence of the above described profile, the contact line length or seal line length between the male and female rotors are reduced. Since the seal line is one of the most important leakage channels inside a helical screw compressor, leading to reduction in both the total efficiency and volumetric efficiency, the reduction of the seal line length has the advantage of increasing performance of the compressor.

As an additional consequence of the above described profile, the drive band between the male and female rotors experience much lower contact stress. For a male drive screw compressor, if point B_M of ellipse segment B_M-C_M is located at the long axis of the ellipse, the radius at point C_M is much larger than the radius at point B_M due to the geometrical feature of an ellipse. The drive band is located on B-C and near point C, and the larger radius results in a larger relative radius, which results in lower contact stress. For a female drive screw compressor, the profile section design of segment F-E also gives the profile the ability to control the contact stress at the drive band.

Although preferred embodiments of the present invention have been illustrated and described, other changes will occur to those skilled in the art. It is therefore intended that the scope of the present invention is to be limited only by the scope of the appended claims.